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Stream Analysis of a 10 Ton Horse River.

A Thesis Submitted to

The Faculty of

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THESIS

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PREFACE.

This thesis was not undertaken in a spirit of criticism or to find fault with the design or construction of the machine, but rather that the writers might become more familiar with the investigation and analysis of stress in machines. This has been the main object throughout the work.

We take this opportunity to thank Professor J.F. Nelson, Professor L.N. Field, Professor C.A. Melick, and others for the aid they have given to us in working this thesis.

OBJECT.

The object of this thesis was to investigate and study the stresses that are produced in the various members of the machine, when the machine is operating under rated load.

The stresses were analyzed and checked by three different solutions: namely;

- (a) graphical solution,
- (b) analytical solution,
- (c) actual test.

Of these three methods, the results from the analytical solution were accepted as correct, having been checked in three positions by the graphical solution.

HISTORY OF THE HANNA FLUID PRESSURE RIVETING MACHINE.

The Hanna type of riveter was first patented by Elmer S. Hanna, of Chicago, Illinois. On May 15, 1901, Mr. Hanna made application for patents on two types of linkages for "fluid pressure riveting machine". These patents were granted May, 1901.

To give an idea of the main claim Mr. Hanna filed in his first application, the following is a copy of Claim No. 1, from the U.S. Patent Office Official Gazette of June 18, 1900.

"Claim No. 1. In a riveting machine, the combination with a supporting frame, having a sliding plunger therein, of means for moving said plunger to a relatively rapid and suddenly reverse movement and for controlling the final movement of a fluid-tight cylinder connected with said means including a main lever, fulcrumed in the frame above the plunger, a link connection between one end of said lever and the plunger, a link pivoted to the free end, a link connection between said link and the opposite end of said lever, and means for moving the link pivoted to the frame about its pivot."

This claim was followed by three other claims of the same tenor, but containing different uses and slight changes in linkages. Claim No. 1, as used by Mr. Hanna's first intentions were for a air clay levering riveter and that by a peculiar combination of levers he managed to reduce the plunger speed from a rapid to a slow action during the same stroke.

From Claim No. 1, of his second application, it can be seen that he also intended a full levering machine as shown

By the following position: run the same course as the above.

"Claim No. 1, In a shell-pressure riveting machine, the combination of a frame, a plunger bearing a riveting or similar tool, a main operating lever, a spring balance for such lever, a link mechanism connecting the main operating lever and plunger, together, a fluid pressure cylinder, a reciprocating piston therein provided with a piston rod projecting therefrom, and an intermediate lever pivotally mounted upon and secured to the frame of the machine and pivotally connected with the piston rod and main operating lever, substantially as described."

This main claim is supplemented by fourteen sub-claims as to the operation and use of the machine.

After a few years of experimenting, Mr. Harris finally reached the conclusion that the action desired in the plunger could be best obtained by combining a toggle action with the levering action. So on June 12, 1910 he filed application for a patent combining the toggle action with the lever action, as shown in Claim No. 1, from the Patent Office Gazette of May 2, 1911.

"Claim No. 1, In a riveting machine, the combination with a pair of dies, of means for relatively moving said dies towards and away from each other, comprising toggle links, an operating lever to which are concentrically pivotally connected the adjacent end of the toggle links, a guide link pivotally connected to said lever and means for applying power to said lever."

This combined toggle and lever is the one used on the modern House Air riveter. The position and combination of the linkages can be seen on Drawing No. 15.

A graphic analysis of the machine shows that there are two distinct mechanical actions during the piston stroke, which are

- (a) A toggle action during the first half of the piston stroke;
- (b) a lever action for the balance or last half of the piston stroke.

The total plunger stroke is 4 inches, of which $3\frac{1}{2}$ inches is taken up by the toggle action and $\frac{1}{2}$ inch by the lever action. At the beginning of the levering action, maximum pressure is reached and continues throughout the entire lever stroke. This brings the plates together and forms the rivet head under maximum pressure.

In the following investigation, the main object was to determine the variation of stress in the several members at various positions during both the levering and toggling actions.

CONSTRUCTION.

The frame of the riveter is a steel casting of the box type, which reduces to a minimum the frame deflection, as well as, the torsion and side thrust of the frame under pressure. The main lever, pushlinks, upper toggles, charger, piston rod head and some other parts are all steel castings, while the main frame bushing and cylinder parts are chiefly of cast iron. The lower toggle, die screw, lower die holder, dies, and piston rod are of a special high carbon steel.

All working joints are bushed with a special aluminum bronze, called by its trade name, "Loren Bronze". The lower toggle also works in heavy removable sockets of the same material.

All bearings pins are made of cold rolled steel. They are drilled and threaded in the center for forcing heavy grease into the bearings.

GRAPHICAL SOLUTION.

The graphical solution for the stress in the various members of the machine was drawn out completely for only three points. The real object of the graphical solution was to furnish a check for the analytical solution, which was calculated from data obtained by means of the graphical layout of the same at positions.

The total travel of the center of the piston pin was first determined, and was found to be 17.375 inches. This was divided into ten parts and the eleven points were laid off on the center line of the cylinder a distance 1.6475" apart. These were then numbered from 1 to 11 inclusive. This gave all positions of the piston from which the stress in the members was determined. The kinematic layout of the relative positions of the joints at each successive piston position was first drawn out, by the aid of a complete set of blue prints of the machine furnished by the Valves Engineering, Sales Company of Chicago. This determined the direction of the several links and the position of the linkage system at the various piston positions.

Then for each position of the mechanism the directions of the force lines in each link was determined by the aid of the friction circles. The driving force was delivered along the connecting rod into the piston. This force was continued in the force polygon with the force lines in the several links, and by this method the stress in all members was graphically determined.

FRICTION CIRCLES

In journal friction the amount of power lost by friction is very small, especially with slow-moving bearings. The resistance from the journal bearings in any case may be resolved into two forces, one the normal pressure on the pin, and the other, the frictional resistance at the circumference of the journal. Then in whatever position the link may be, the two resistances will be met with. Therefore the directions and the amount of the frictional resistances are determined by the tangent to a friction circle the radius of which is always;

$$\rho = r \sin \phi$$

Where

$$\rho = \text{radius of friction circle}$$

$$r = \text{radius of journal}$$

$$\phi = \text{angle whose tangent is}$$

$$\mu = \text{co-efficient of friction between the two metallic surfaces: namely, the pin and the journal.}$$

The bearing materials used in the Home Governor are cast-iron and brass and all parts are cold-rolled steel. They are slow-moving bearings subjected to very high pressures. This point only needs, because of the great pressure, any lubrication with metal surfaces. From Kirk's Mechanical Engineering Hand Book, Page 21, Table VI, 11a, and Kirk 11 and Perry's Elements of Machine Design, Page 11, a value for the co-efficient of friction for these conditions was taken at 0.15

$$\text{Then } \mu = 0.15 = \tan \phi$$

$$\text{Hence } \phi = \text{arc sine } 0.15 = 8.5^\circ$$

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"Claim No. 1. In a riveting machine, the combination with a supporting frame, having a sliding plunger therein, of means for moving said plunger at a relatively rapid and gradually decreasing speed and for controlling the final movement at a substantially uniform speed and maximum pressure, said means including a main lever, fulcrumed on the frame above the plunger, a link connection between one end of said lever and the plunger, a link pivoted to the free end, a link connection between said link and the opposite end of said lever, and means for moving the link pivoted to the frame about its pivot."

This claim was followed by three other claims of the same character, but containing different words and slight changes in linkages. Claim No. 1, shows that Mr. Hanna's first intentions were for a strictly leveraging riveter and that by a peculiar combination of levers he sought to reduce the plunger speed from a rapid to a slow action during the same stroke.

From Claim No. 1, of his second application, it can be seen that he also intended a roll leveraging machine as shown

By the following notation from the same source as the above.

"Claim No. 1, In a fluid-pressure riveting machine, the combination of a frame, a plunger bearing a riveting or similar tool, a main operating lever, a springing fulcrum for such lever, link mechanism connecting the main operating lever and plunger together, a fluid pressure cylinder, a reciprocating piston therein provided with a piston rod projecting therefrom, and an intermediate lever pivoted upon and secured to the frame of the machine and pivotally connected with the piston rod and main operating lever, substantially as described."

This main claim is supplemented by fourteen sub-claims as to the operation and use of the machine.

After a few years of experimenting, Mr. Hawk finally reached the conclusion that the action desired in the plunger could be best obtained by combining a toggle action with the levering action. So on June 10, 1905 he filed application for a patent combining the toggle action with the lever action, as shown in Claim No. 1, from the Patent Office Gazette of May 2, 1911.

"Claim No. 1, In a riveting machine, the combination with a pair of dies, of means for relatively moving said dies towards and away from each other, comprising toggle links, an operating lever to which are eccentrically pivotally connected the adjacent end of the toggle links, a guide link pivotally connected to said lever and means for applying power to said lever."

This combined toggle and lever is the one used on the modern Henry Air riveter. The position and configuration of the linkage can be seen on drawing No. 15.

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In the following investigation, the main object was to determine the variation of stress in the several members at various positions during both the levering and toggling actions.

CONSTRUCTION.

The frame of the riveter is a steel casting of the box type, which reduces to a minimum the frame deflection, as well as, the torsion and side thrust of the frame under pressure. The main lever, guide links, upper toggles, chumger, piston rod head and connecting parts are all steel castings, while the main frame bushing and cylinder parts are chiefly of cast iron. The lower toggle, die screw, lower die holder, die, and piston rod are of a special high carbon steel.

All working joints are bushed with a special aluminum bronze, called by its trade name, "Lucan Bronze". The lower toggle also works in heavy removable sockets of the same material.

All bearings pins are made of cold rolled steel. They are drilled and threaded in the center for forcing heavy grease into the bearings.

GRAPHICAL SOLUTION.

The graphical solution for the stress in the various members of the machine was drawn out completely for only three points. The real object of the graphical solution was to furnish a check for the analytical solution, which was calculated from data obtained by means of the graphical layout of the several positions.

The total travel of the center of the piston pin was first determined, and was found to be 12.375 inches. This was divided into ten parts and the eleven points were laid off on the center line of the cylinder a distance 1.2375" apart. These were then numbered from 1 to 11 inclusive. This gave all positions of the piston from each of which the stress in the members was determined. The kinematic layout of the relative positions of the joints at each successive piston position was first drawn out, by the aid of a complete set of blue prints of the machine furnished by the Valves Engineering, Sales Company of Chicago. This determined the direction of the several links and the position of the linkage system at the various piston positions.

Then for each position of the mechanism the direction of the force lines in each link was determined by the aid of the friction circles. The driving force was delivered along the connecting rod from the piston. This force was combined in the force polygon with the force lines in the several links, and by this method the stress in all members was graphically determined.

FRICTION CIRCLES

In journal friction the amount of power lost by friction is very small, especially with slow-moving bearings. The resistance from the journal bearings in any case may be resolved into two forces, one the normal pressure on the pin, and the other, the frictional resistance at the circumference of the journal. Then in whatever position the line may be, the same frictional resistance will be met with. Therefore the direction and the amount of the frictional resistance are determined by the tangent to a friction circle the radius of which is always:

$$\rho = r \sin \phi$$

Where

ρ = radius of friction circle

r = radius of journal

ϕ = angle whose tangent is

μ = coefficient of friction between the two metallic surfaces: namely the pin and the journal.

The bearing metals used in the Harney Divisor are of the 12 in. diameter and all parts are cold-rolled steel. They are slow-moving bearings subjected to very high pressures. This practically means, because of the great pressure, dry lubrication with metal surfaces. From Merritt's Mechanical Engineering Hand Book, Page 11, Friction Tables, and Kibbrell and Merritt's Elements of Machine Design, Page 11, the average coefficient of friction for these conditions was taken as .13

$$\text{Then } \mu = .13 = \tan \phi$$

$$\text{and for } \phi = \text{arc sine } .13 = 7.5^\circ$$

Therefore $\sin \phi = .014000$

From the formula given above, the radius of the friction circles of the bearings in this machine is

$$\rho = r \sin \phi = r \times .014000$$

For example:

Radius of the Upper Toggle pin = 1.175 inches.

Then r = 1.175 inches.

Then ρ = 1.175 x 0.014000

Or ρ = .01645 inches.

The same formula was applied to all the pins, with the following results:

Position of Pin	Radius of Pin	Radius of Friction Circle
Upper Toggle Upper Pin	1.175"	0.016"
Lower Toggle Pin (both)	1.175"	0.016"
Upper Toggle Lower Pin	1.500"	0.021"
Wrist Pin	0.750"	0.011"
Piston Rod Head Pin	.750"	0.011"
Cutler Link Upper Pin	0.750"	0.011"
Cutler Link Lower Pin	0.875"	0.012"

These various friction circles were then drawn in around their proper centers, and the direction of rotation of the bearing, at the pin, was determined, and was indicated by a small arrow on the outside of the centers of the member in question. This is shown on the graphical solution on Drawing No. 34.

The side of the tangency of the force line to these circles was determined by the character of the stress in the members, that is, whether they were in tension or compression. When in tension, the lines were drawn tangent to the friction circles on the side opposite the direction of motion and when in compression, on the same side as the direction of motion. From this it can readily be seen that the effect of the friction circles was merely to change the direction of the force lines enough to compensate in the graphical solution for the amount lost in the friction in the journal.

EFFECTIVE PRESSURE IN CONNECTING ROD PRESSURE LINE.

The pressure effectively delivered to the connecting rod pressure line from the total force developed against the piston head, is based on two considerations:

- (a) the angularity of the rod;
- (b) the friction in the cylinder.

The graphical solution of this can best be shown by the sketch on the Drawing No. 1.

Suppose P = the total pressure on the cylinder head then P = area of piston head times the air pressure. Since the piston motion is confined by the cylinder, P acts along the center line of the cylinder. However, the connecting rod force line acts at an angle with the center line of the cylinder. Thus the force P is resolved into two components: one, in the direction of OX , along the force line; and the other, MN , perpendicular to it.

When P is laid off to a definite scale, OM is the force along the center line of the piston travel. ON is the component of OM along the connecting rod line, while EN is the perpendicular component to OM . The pressure ON along the defined line would be the effective pressure, if it were not for the friction of the piston in the cylinder. This acts at an angle whose tangent is μ , the co-efficient of friction. As in the case of the journals, μ was assumed to be 0.15.

$$\text{Then } \mu = \tan \phi$$

$$= 0.15$$

$$\tan \phi = 0.15$$

$$\text{Arc } \tan 0.15 = 8^{\circ}32'$$

This angle was laid off from P on the center line of the cylinder, with the angle between the line and the perpendicular EN . This friction line cut the resultant pressure line at A , which gives OA as the effective pressure delivered. The side of the cylinder on which the friction angle is effective depends on the angularity of the connecting rod, whether it is above or below the center line of the cylinder.

TOTAL PRESSURE DELIVERED TO THE PIR BLOCK.

This pressure was determined from the pressure already calculated in the lower toggle, and depended upon two considerations:

- A. the angularity of the toggle with respect to the center line of the plunger;
- B. the friction in the plunger.

Because of the long bearing surface, the friction tends to reduce the effective pressure at both ends of the plunger. Thus the friction components act in this relation.

The points of application of the friction components were assumed to be at $1/3$ of the total bearing length, from the edge of the bearing. The friction lines were drawn at the angle whose tangent is μ . These combined with the pressure in the lower toggle and the die block pressure gave the effective pressure on the die block.

Tabulated Results

Scale $1" = 7,000\psi$

ANALYTICAL SOLUTION

The analytical solution was based upon the graphical solution in three ways, namely, -

- (a) The pressure in the piston rod;
- (b) The direction of pressure in the links;
- (c) Perpendicular distances from intersection points.

The pressure in the piston rod was determined by subtracting, graphically, the loss in pressure due to the friction of the piston walls and also the reduction due to the angularity of the piston rod.

The direction of pressure in the links was not coincident with the center line, theoretically, owing to the friction of the journals. Hence the pressure lines used were the same in direction as used in the graphical solution.

The Main Lever acts as a tie block for the four links and hence we do not direct our attention to the forces in it, but to the forces in the links.

There were then four forces acting, three of which were unknown. The general method of solution was to take the intersection of any two of the unknown forces as a center, and determine the third by taking moments about the assumed center.

In Drawing No. 2, line R_4 represents the direction of the force in the upper toggle: R_3 , the direction of the force in the lower toggle: R_5 the direction of the force in the guide link. Also,

- (P) is the pressure in the piston rod.
- (M) is the intersection between the upper and lower toggles.

(N) is the intersection between the lower toggle and the guide link

(O) is the intersection between the guide link and the piston rod.

(T) is the intersection between the guide link and the upper toggle.

(W) is the intersection between the lower toggle and the piston rod.

(a) is the perpendicular distance from (M) to force (R_5)

(b) " " " " " (M) " " (P)

(c) " " " " " (N) " " (R_4)

(d) " " " " " (N) " " (P)

(e) " " " " " (O) " " (R_2)

(f) " " " " " (C) " " (R_4)

(g) " " " " " (T) " " (R_2)

(h) " " " " " (T) " " (P)

(i) " " " " " (W) " " (R_5)

(j) " " " " " (W) " " (R_4)

In all positions except No.6, moments were first taken about point (M).

$$R_5 \times a = P \times b \quad \text{or} \quad R_5 = \frac{P \times b}{a}$$

Next, moments were taken about (N).

$$R_4 \times c = P \times d \quad \text{or} \quad R_4 = \frac{P \times d}{c}$$

Finally, moments were taken about (O).

$$R_2 \times e = P \times f \quad \text{or} \quad R_2 = \frac{P \times f}{e}$$

In position No.6, the action of the links was changing from a toggling to a levering action and the force line of the

the upper and lower toggles were so nearly parallel as to make the intersection (M) inconvenient. Hence the solution was made in the following order.

First, moments were taken about (T).

$$R_2 \times g = P \times h \quad \text{or} \quad R_2 = \frac{P \times h}{g}$$

Secondly, moments were taken about (C).

$$R_2 \times e = R_4 \times f \quad \text{or} \quad R_4 = \frac{R_2 \times e}{f}$$

Lastly, moments were taken about (W).

$$R_5 \times i = R_4 \times k \quad \text{or} \quad R_5 = \frac{R_4 \times k}{i}$$

[illegible]

• (c) *the date of the last day of the year*

[illegible]

ANALYTICAL SOLUTION FOR DIE BLACK METHOD.

The analytical solution of this problem was obtained from the graphical solution in four steps:

- a. pressure in the die or cylinder;
- b. regularity of the regions in the line of the plunger;
- c. direction and position of friction lines;
- d. distance to intersection of the friction lines.

From the graphical solution of Figure No. 1, the result indicated was to lead to be as follows:

The friction lines that were effective for each position were extended to an intersection at C. This indicated the position since C was taken as a center for a circle.

The perpendicular distance was measured from C to the line called OP. Because the friction angle reached the same, the distance from P to C, or CP, was always constant, leaving the length of the line S_2 and the angle the only variable.

The value of the extent of S_2 around C as a center was determined by measuring the perpendicular distance from C to P_1 .

The formula then becomes;

$$P \times CP = S_2 \times CR,$$

$$\text{or } P = \frac{R \times CS_2}{CP}$$

From the results found a pressure curve was plotted for the Die Black.

METHOD OF TESTING THE HENCKE PIVETER.

The test was divided into two parts: namely,

(a) test of the Frame.

(b) test of the Links.

TEST OF THE FRAME.

The object of the test was to determine the stress in the frame at different points when pressure was applied, in order to determine the location of the neutral axis in the curved part of the frame.

Drawing No. 31 shows the frame coating. Fracture marks were laid out on both sides of the frame as shown on the drawing. As the pressure is applied to the piston the tendency is to spread the frame apart at the nose. The tendency of the metal, then, would be for the outer fibers to be in compression and the inner ones in tension.

By using a Perry Strain Gage and taking measurements with there was no pressure on the piston, and then when the pressure was on it, we determined the deformation at the different points.

From the data thus obtained we average the deformation at the points in the frame. For example, we assumed that the deformation of $\frac{1.011 + 1.001}{2}$ = average deformation. Thus, after averaging the deformation, we had a series of points which were plotted, with regard to a certain axis, was the locus of a stress line. Where the line crossed the axis, was the neutral point. In other words, the intersection of the stress line and the reference line located the point where there was neither tension nor compression, but zero stress.

We took measurements at three different angles in the sheet, as shown by the drawing, and hence determined three points on the neutral axis.

TEST OF LINKS.

(Including final pressure on the plunger)

The object of testing the links was to determine the stress in them at different positions of the piston so as to furnish a check on our analytical and graphical solution.

The method of determining the stress in the rollers was similar to that used for determining the stress in the links.

The Guide links and the Upper Toggles were pinned, as shown in drawings No. 12 and 13, respectively. By means of the Heavy Bar in Case, zero readings of the links were taken before any pressure was applied. The pressure was then applied, and the deformation of the various links was read off.

We then applied the general formula for all three the Principles of Elasticity.

$$M = \frac{E \cdot \Delta}{L} \quad \text{or} \quad P = \frac{M \cdot L}{\Delta}$$

and $P = \text{Load}$

$M = \text{Modulus of Elasticity}$

$\Delta = \text{Cross-sectional area}$

$L = \text{Length between points of measurement.}$

$\Delta = \text{Deformation}$

The reaction will be in compression or tension depending on the action of the reaction Δ . Hence from the test with the stress in the various rollers was computed.

On account of the close construction of the machine, it was impossible to measure the stress in the lower toggles; as

The following apparatus was constructed to measure the load on the plunger directly.

CALCULATIONS FOR TEST BLOCK.

The determination of the pressure, in this instance dependent upon the elasticity of the metal, was made in the following way: From zero load up until the elastic limit is reached, the deformation is proportional to the load. Hence, by constructing the test block and calibrating it, we could measure the fluid die or plunger pressure by measuring the compression of the block.

The test block, shown in drawing No. 7, was calculated for the maximum pressure which we determined analytically, thus:

Maximum pressure ----- 17,000 lbs.

Elastic limit of steel ----- 30,000 lbs./sq. in.

Area required ----- 3, 11 / 30,000 = 0.00103 sq. in.

Assume an outside diameter of 3 in.

Area of 3 in. circle ----- 7.0686 sq. in.

" " 2 1/2 in. circle ----- 4.9087 " "

Net area ----- 2.1598 " "

To check for the probable unit deformation

$$d = \frac{P \cdot l}{A \cdot E} = \frac{170,000 \times 3}{2.1598 \times 30,000,000} = 0.00407 \text{ in.}$$

Total deformation ---- . 0407 x 3 ----- 0.01221 in.

Since the total deformation as calculated would not give a large enough reading to be accurate, the extensometer frame was made as shown on drawing No. 7.

The frame built for the reading is thus; hence the

total deformation to read on the dial gage would equal;

$$0.01404 \times 5 = 0.0702 \text{ in.}$$

To test the base of the block for shear

$$\text{Shear area} = \pi D t = 3.1416 \times 1.6472 \times 0.75 = 3.87 \text{ sq.in.}$$

Total load -----173,000 lbs.

Ultimate load in shear ----0,000 lbs. / sq.in.

Required area ----173,000/0,000 = 0.41 sq.in.

Actual area -----3.87 " "

Factor of safety ----3.30/0.4 ----1.35

Since the test block has a bearing surface the latter, the calculations are correct.

DATA FOR ANALYTICAL SOLUTION.

No.	P		W		C	
	a	b	c	d	e	f
1	5.69	6.78	.63	4.78	5.51	5.19
2	6.00	6.60	5.22	5.24	6.14	6.10
3	6.12	6.75	3.60	7.26	7.1	6.47
4	6.15	6.38	2.37	2.51	2.63	2.73
5	10.17	3.37	1.44	7.32	12.71	11.37
6					11.71	11.85
7	6.16	18.46	0.13	10.60	13.42	13.32
8	4.78	11.22	0.65	10.6	12.72	14.52
9	4.13	15.15	0.00	11.11	13.00	15.16
10	4.61	15.90	1.08	10.17	11.11	15.63
11	6.05	16.72	1.17	10.42	13.0	15.18

No.	T		T		P
	g	h	i	k	
1					1200
2					225
3					3450
4					2530
5					550
6	0.97	11.35	.98	1.12	2550
7					2530
8					2500
9					250
10					2715
11					2700

RESULTS OF THE ANALYTICAL SOLUTION.

No.	R_1	R_2	R_4	R_5
1	6,400	7,375	8,700	1,200
2	7,000	8,000	9,100	1,100
3	7,600	8,600	9,600	1,000
4	8,200	9,200	10,200	900
5	8,800	9,800	10,800	800
6	9,400	10,400	11,400	700
7	10,000	11,000	12,000	600
8	10,600	11,600	12,600	500
9	11,200	12,200	13,200	400
10	11,800	12,800	13,800	300
11	12,400	13,400	14,400	200

DATA AND RESULTS: FOR THE GRAPHICAL SOLUTION.

Air pressure----- 114.5 psia.

Force pressure along piston center line----- 110.7.

Scale----- lin. 75 = 1.

PISTON ROD.

The pressures in the piston rod were all compressive
as follows.

Position	Nature of stress	Length	Stress
1	Compression	1.74"	81,000
2	Compression	1.845"	82,150
3	Compression	1.89"	83,300
4	Compression	1.71"	80,000
5	Compression	1.71"	80,000
6	Compression	1.71"	80,000
7	Compression	1.71"	80,000
8	Compression	1.88"	81,000
9	Compression	1.85"	80,000
10	Compression	1.845"	80,150
11	Compression	1.84"	80,100

GUIDE LINKS.

Position	Nature of stress	Line number	length	stress
	Compression	C ₁ E ₁	1.1 "	100,000
6	Tension	C ₆ B ₆	1.51"	11,500
11	Tension	C ₁₁ B ₁₁	3.0 "	11,500

UPPER TOGGLES.

Position	Nature of stress	Line number	Length	Stress
5	Compression	A B ₁	1.24"	153,154
6	Compression	A ₆ B ₆	17.47"	153,154
11	Compression	A ₁₁ B ₁₁	1.71"	153,154

LOWER TOGGLES.

Position	Nature of stress	Line number	Length	Stress
5	Compression	C ₁ B ₅	1.15"	157,755
6	Compression	C ₆ B ₆	15.85"	157,755
11	Compression	C ₁₁ B ₁₁	11.27"	157,755

DIE BLOCK PRESSURE.

Position	Nature of stress	Line number	Length	Stress
5	Compression	C ₂ D ₁	1.35"	161,854
6	Compression	E ₆ F ₆	15.47"	161,854
11	Compression	E ₁₁ F ₁₁	11.07"	161,854

DATA OF ACTUAL TEST.

Test Block Calibration.

Die Block Pressure.

Load Lbs.	Deflection. 1/1000 in.	Position Piston inches	Deflection 1/1000 in.	Load Lbs.
500	0.0	11 3/8	1	500
1000	1.1	11 1/2	3	1000
1100	2.0	11	4	1100
1200	3.4	11 1/4	5	1200
1300	4.5	11 1/2	11	1300
1400	5.0	11	10.5	1400
1500	5.9	11 3/8	10.3	1500
1600	6.8	11 3/8	10.0	1600
1700	7.4	11 3/8	9.9	1700
1800	8.0	11 1/4	9.7	1800
1900	8.5	11 3/8	9.5	1900
2000	10.0	11	9.3	2000
2100	10.7			
2200	11.5			
2300	12.3			
2400	13.0			
2500	14.2			
2600	15.5			

CONCLUSION

In conclusion, the writers wish to compare the results obtained and recorded in their work, and attempt to point out the reasons for certain discrepancies which seem to appear in time.

We found that by the analytical and graphical solutions we had a varying pressure on the die block, as shown in Drawing No. 3. Also, the analytical and graphical solutions showed us that the maximum pressure was reached a little beyond the center strike. It was noted that at the center of the strike, the die block action changed from a toggling to a levering action. It is the opinion of the writers that the sudden rise of pressure observed in the curve (Drawing No. 3) was due to the die block going from lever action. The force lines on the toggles at these points are nearly parallel.

Due to unavoidable and unforeseen delays and interruptions, the actual test of the machine was not so thorough or complete as was desired by the writers. However, it proved to us that the die block pressure curve had much the same general shape as was found by the analytical solution. The comparison of the test is found on Drawing No. 3.

There seems to be a large discrepancy in either the calculations or the test results. It should be noted that there were several, seriously untrue, assumptions made in the construction of the analytical and graphical solutions. There was no way, except checking by actual test, of knowing

whether the different parts were temporarily sprung because of the great pressure.

The writers believe that the seeming discrepancy was due to the fact that in the analytical and gravimetric calculations, no accurate oil weight could be made for the spring of the frame. It is therefore very possible that the excess pressure found in the analytical section was taken up in the water test, by the spring of the frame.

The strains in the curved frame and their analysis would be a theme in itself, and the writers did not have the opportunity to check the strains in the water frame as was outlined and planned on page No. 12.

DRAWINGS

RESULTS OF THE ANALYTICAL SOLUTION.

No.	F	F ₂	F ₄	F ₃
1	0,000	0,000	0,000	0,000
2	1,000	0,000	0,000	0,000
3	2,000	0,000	0,000	0,000
4	3,000	0,000	0,000	0,000
5	4,000	0,000	0,000	0,000
6	5,000	0,000	0,000	0,000
7	6,000	0,000	0,000	0,000
8	7,000	0,000	0,000	0,000
9	8,000	0,000	0,000	0,000
10	9,000	0,000	0,000	0,000
11	10,000	0,000	0,000	0,000

DATA AND RESULTS FOR THE GRAPHICAL SOLUTION.

Air pressure----- 100 lb. sq. in.

Exhaust pressure along piston center line----- 14.0 lb. sq. in.

Stroke----- 1 in. 75 %.

PISTON ROD.

The pressures in the piston rod were not computed graphically.

Position	Nature of stress	Length	Stress
1	Compression	1.74"	81,000
2	Compression	1.845"	82,500
3	Compression	1.89"	83,000
4	Compression	1.71"	80,000
5	Compression	1.71"	80,000
6	Compression	1.71"	80,000
7	Compression	1.71"	80,000
8	Compression	1.68"	81,000
9	Compression	1.65"	80,000
10	Compression	1.845"	82,500
11	Compression	1.84"	82,000

GUIDE LINKS.

Position	Nature of stress	Line number	Length	Stress
5	Compression	C ₅ E ₅	1.12"	11,000
6	Tension	C ₆ B ₆	1.31"	12,500
11	Tension	C ₁₁ B ₁₁	3.4"	31,000

UPPER TOGGLES.

Position	Nature of stress	Line number	Length	Stress
2	Compression	A ₁ B ₁	1.24"	153,114
6	Compression	A ₅ B ₅	17.47"	153,114
11	Compression	A ₁₁ B ₁₁	12.71"	153,114

LOWER TOGGLES.

Position	Nature of stress	Line number	Length	Stress
2	Compression	C ₁ D ₁	1.15"	101,251
6	Compression	C ₅ D ₅	15.35"	101,251
11	Compression	C ₁₁ D ₁₁	11.57"	101,251

DIE BLOCK PRESSURE.

Position	Nature of stress	Line number	Length	Stress
2	Compression	E ₁ F ₁	1.15"	101,251
6	Compression	E ₅ F ₅	15.47"	101,251
11	Compression	E ₁₁ F ₁₁	11.57"	101,251

DATA OF ACTUAL TEST.

Test Block Calibration.

Die Block Pressure.

Load Lbs.	Deflection. 1/1000 in.	Position Piston inches	Deflection 1/1000 in.	Load Lbs.
000	0.0	11 7/8	1	000
1000	1.5	11 1/8	3	1500
11000	2.0	11	4	2000
12000	3.4	11 1/4	5	2500
21000	4.7	11 1/2	11	3000
30000	5.0	11	10.3	3500
31000	5.0	11 1/2	10.3	4000
40000	6.2	11 2/3	11.3	4500
41000	7.4	11 3/4	11.5	5000
45000	8.0	11 1/4	13.2	5500
51000	9.3	11 3/4	14.2	6000
55000	10.0	11	14.5	6500
61000	12.7			
65000	11.5			
71000	12.3			
75000	13.0			
81000	14.2			
85000	14.5			

CONCLUSION

In conclusion, the writers wish to compare the results obtained and recorded in their work, and attempt to point out the reasons for certain discrepancies which seem to appear in them.

We found that by the analytical and graphical solutions we had a varying pressure on the die block, as shown in Drawing No. 3. Also, the analytical and graphical solutions showed us that the maximum pressure was reached a little beyond the center stroke. It was noted that at the center of the stroke, the kinematic action changed from a toggling to a levering action. It is the opinion of the writers that the sudden rise in pressure observed in the curve (Drawing No. 3) was due to the combined toggle and lever action. The force lines on the toggles at their joints are nearly parallel.

Due to unavoidable and unforeseen delays and interruptions, the actual test of the machine was not so thorough or complete as was desired by the writers. However, it proved to us that the die block pressure curve had much the same general shape as was found by the analytical solution. The comparison of the two is found on Drawing No. 3.

There seems to be a large discrepancy in either the current theory or the test results. It should be noted that there were several, necessarily untrue, assumptions made in the construction of the analytical and graphical solutions. There was no way, except checking by actual test, of knowing

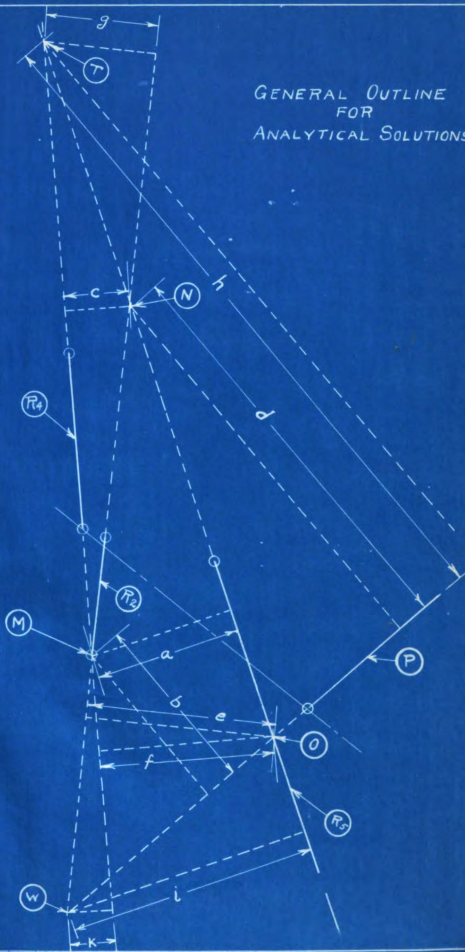
whether the different parts were temporarily sprung because of the great pressure.

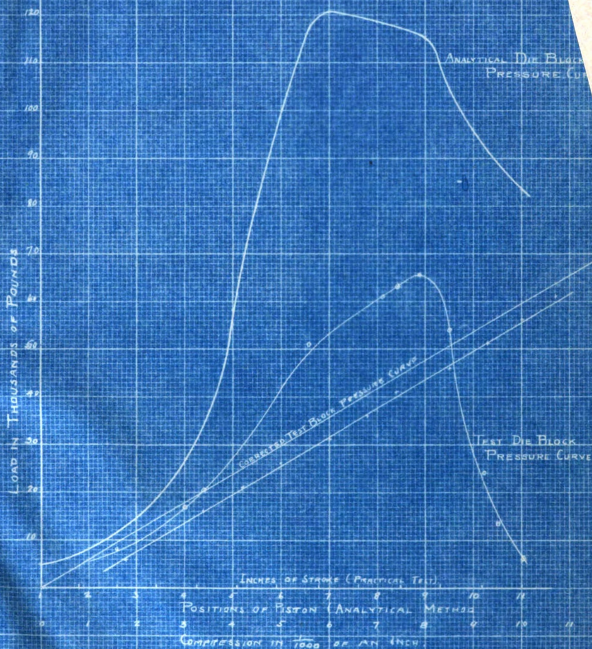
The writers believe that the seeming discrepancy was due to the fact that in the analytical and graphical solutions, to secure all wires could be made for the spring of the frame. It is therefore very possible that the excess pressure found in the analytical solution was taken up in the actual test, by the spring of the frame.

The stresses in the curved frame and their analysis would be a theme in itself, and the writers did not have the opportunity to check the stresses in the Flywheel frame as was outlined and planned on page No. 22.

DRAWINGS

GENERAL OUTLINE
FOR
ANALYTICAL SOLUTIONS

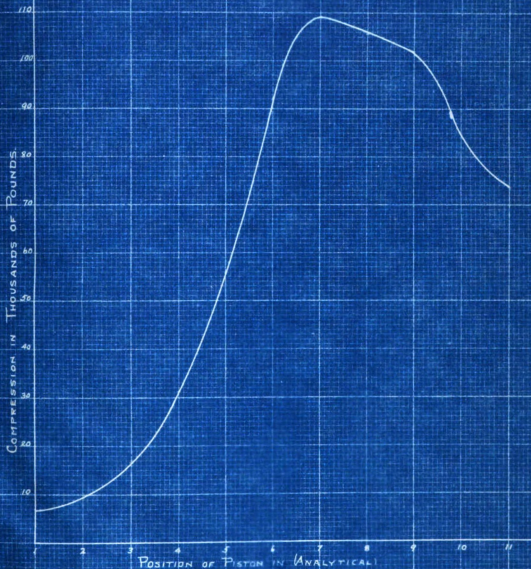




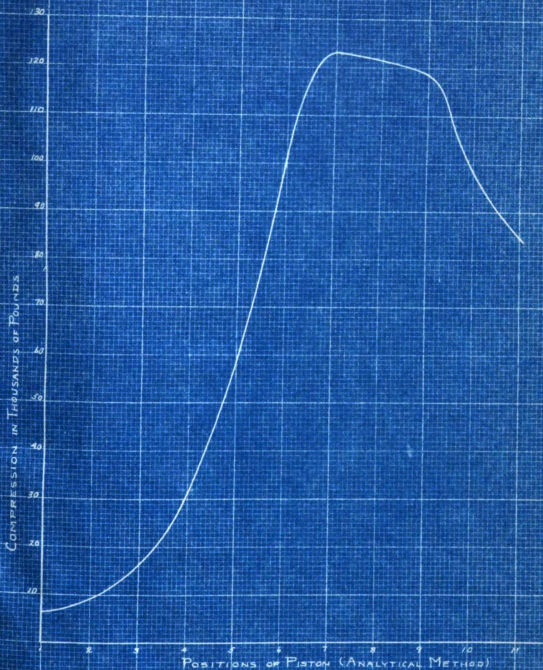
DIE BLOCK PRESSURE CURVES

THESIS OF
H. E. MORTON N. E. MARCH

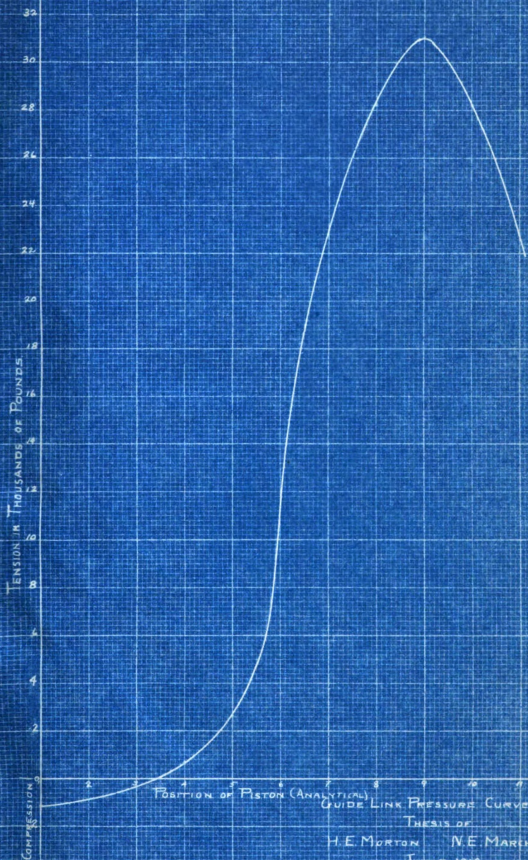
JUNE 1916

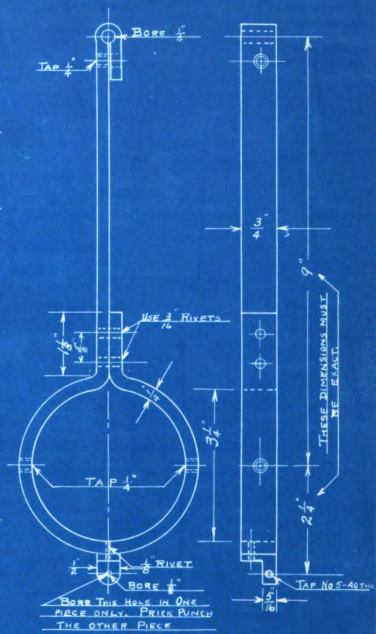
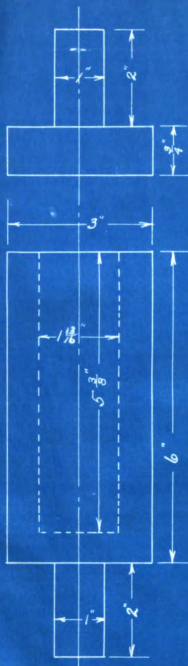


UPPER TOGGLE PRESSURE CURVE
 THESIS OF
 H.E. MORTON N.E. MARCH
 JUNE 1916



LOWER TOGGLE PRESSURE CURVE
THESIS OF
H. E. MORTON N. E. MARCH
JUNE 1916





ONE REQUIRED CRS.
TEST BLOCK

2 REQUIRED CRS.

HANNA 30 T. RIVETER
TESTING APPARATUS
THESIS OF
H.E. MORTON N.E. MARCH
M.A.C.
JUNE 1916







← .875" →

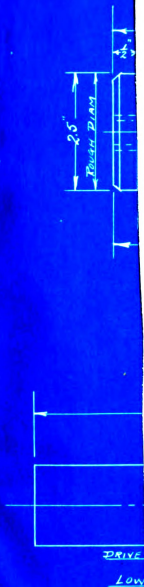
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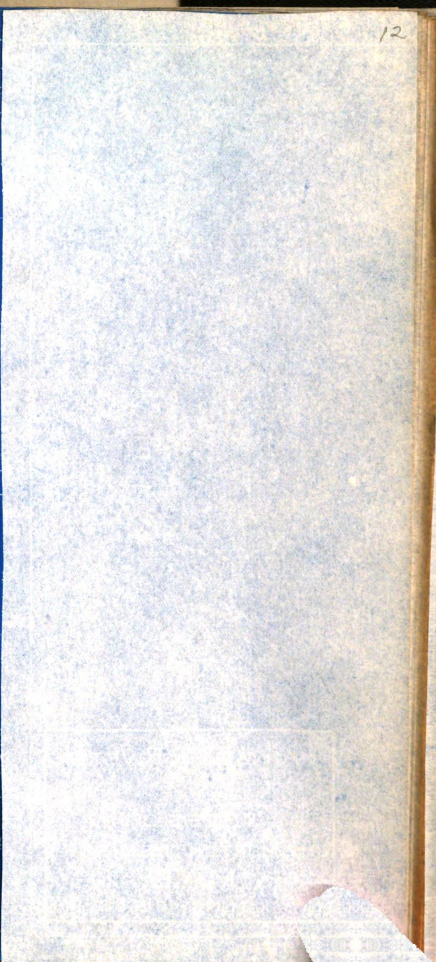
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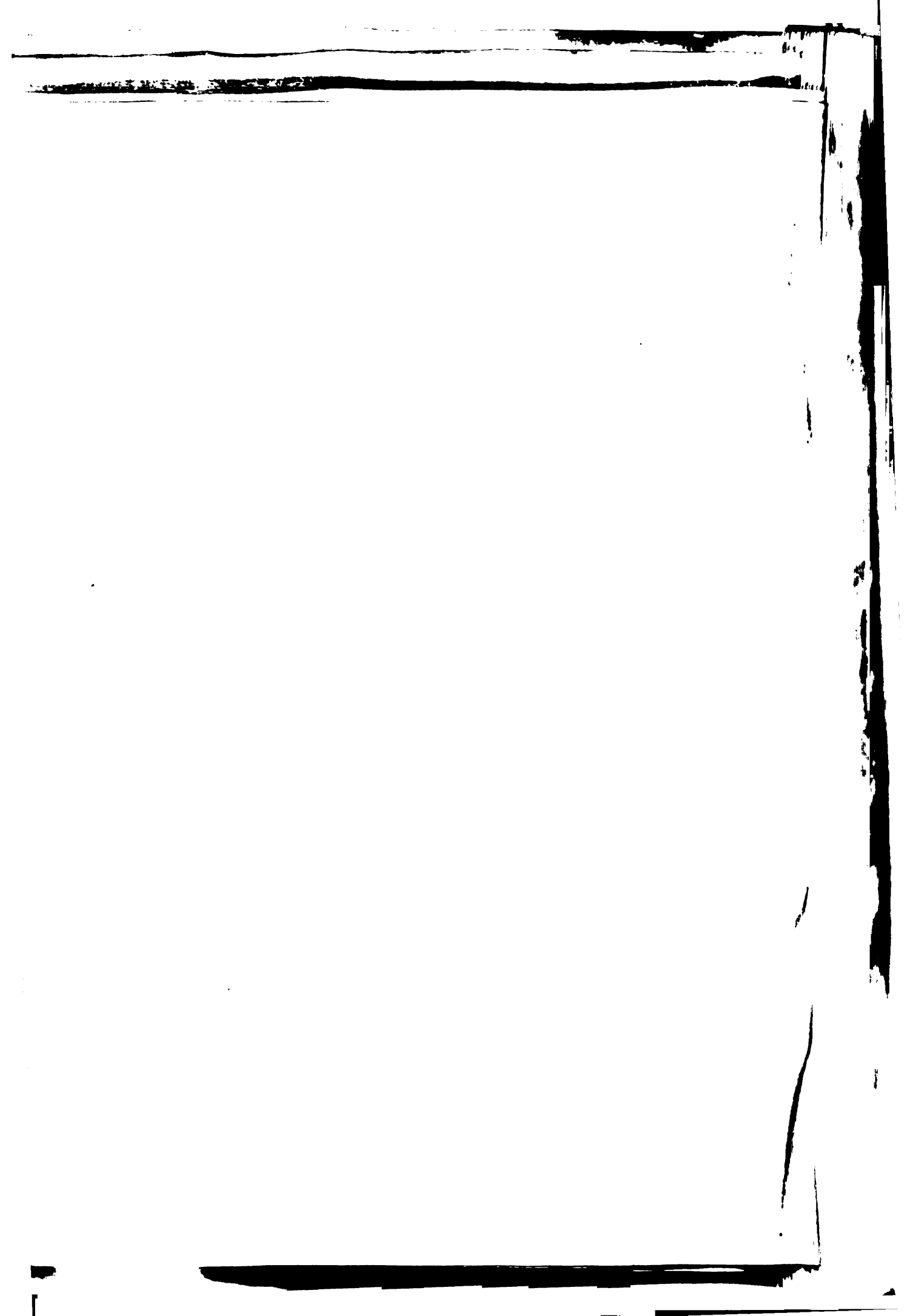
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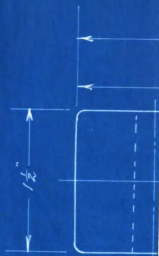
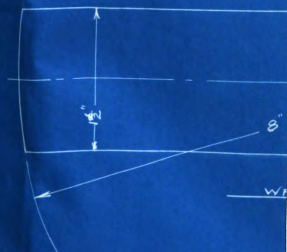
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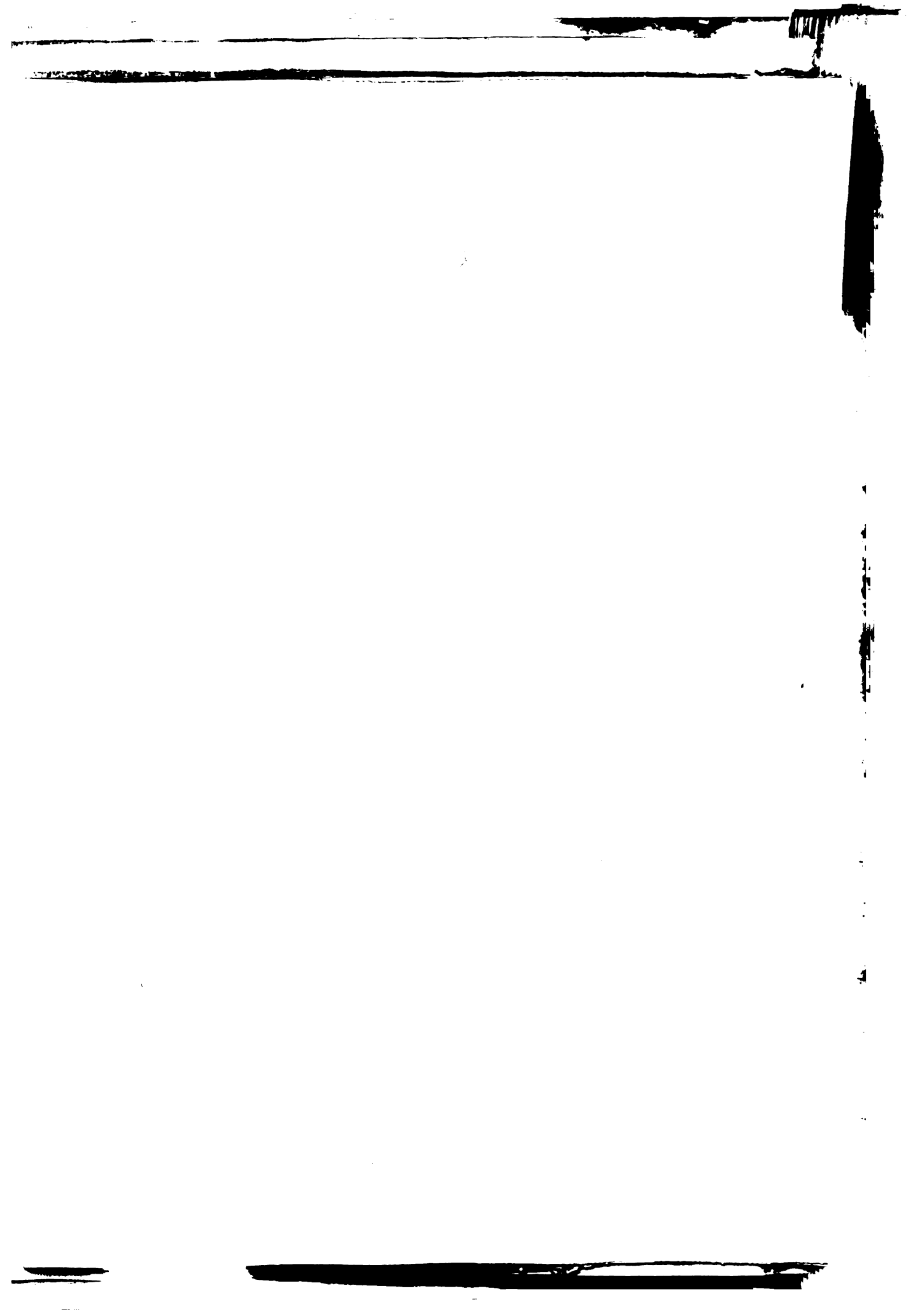
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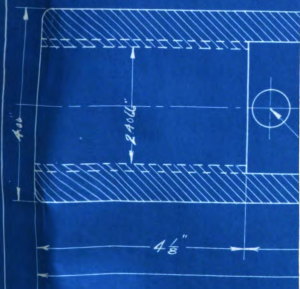


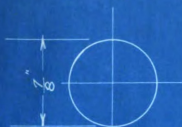




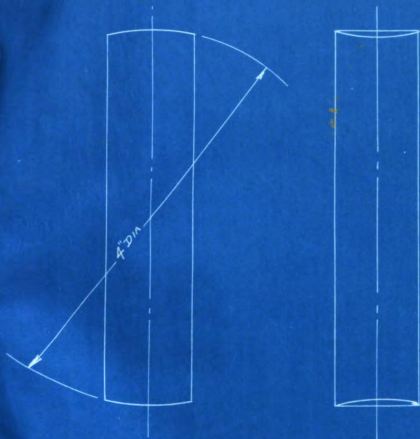








DRIVE FIT IN PLUNGER.



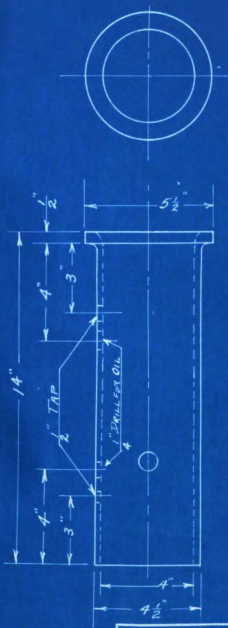
PLUNGER PIN

HANNA S. T. RIVETER
PLUNGER PIN
THESIS OF
H. E. MORTON N. E. MARCH
M.A.C.
JUNE 1916









HANNA 30 T. RIVETER

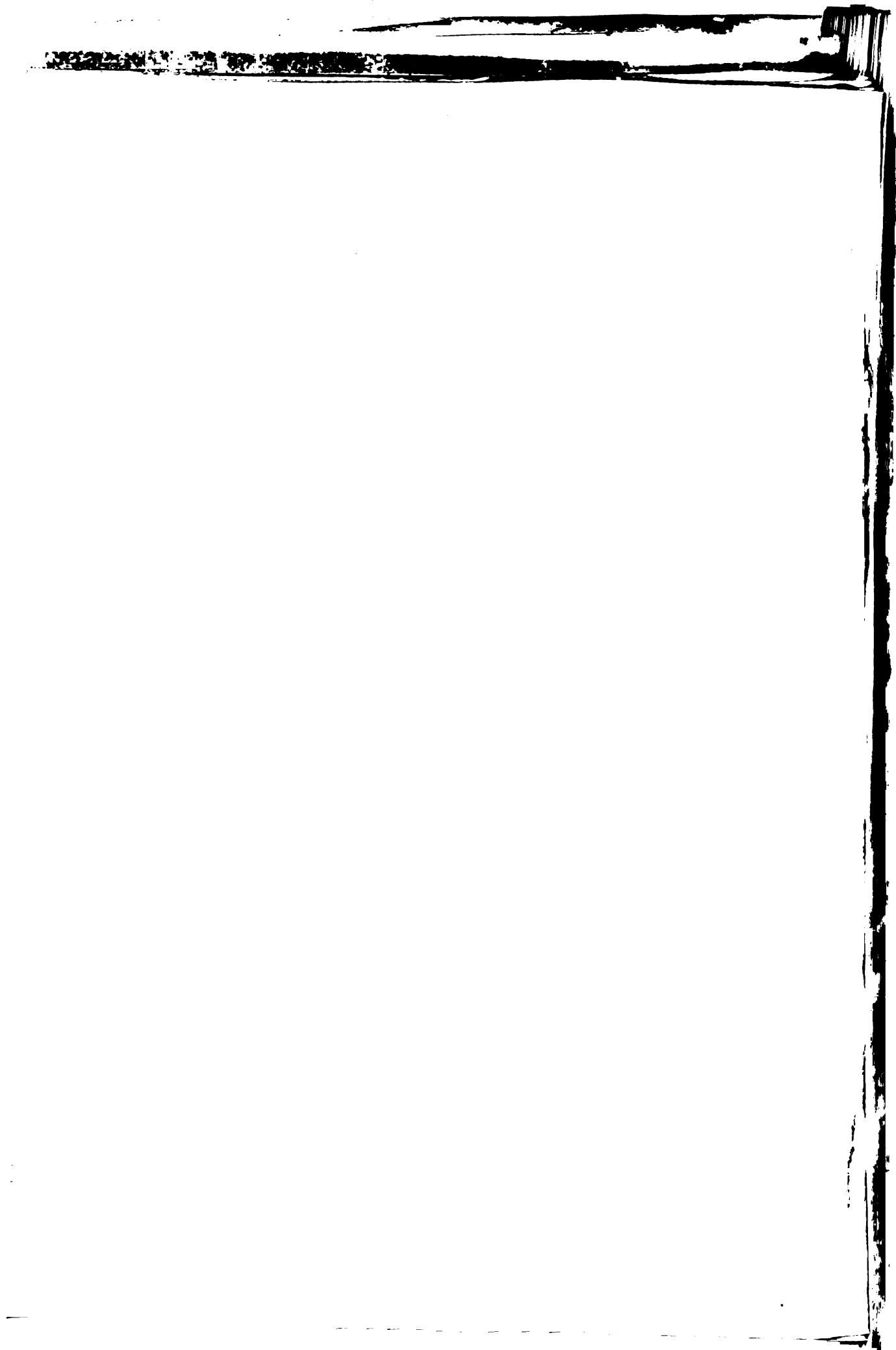
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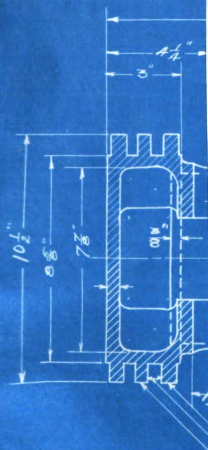
THE SIS OF

H.E. MORTON NE. MARCH

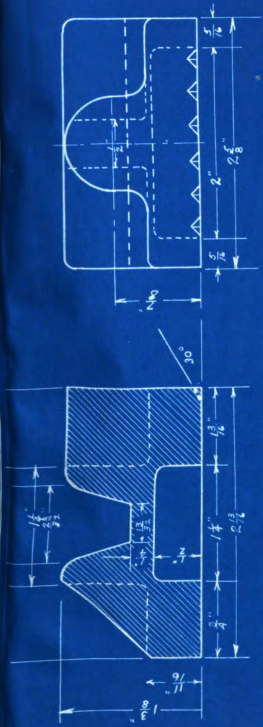
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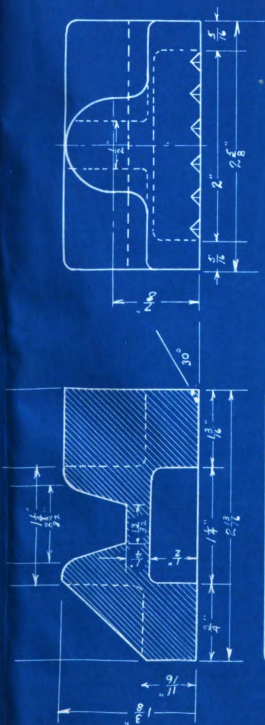




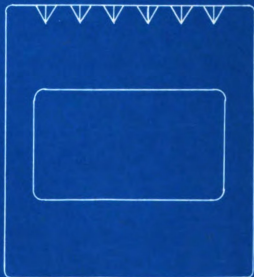


HANNA 30 T. RIVETER
VALVE
THEESIS OF
HE. MORTON NE. MARCH
M.A.C.
JUNE 1916





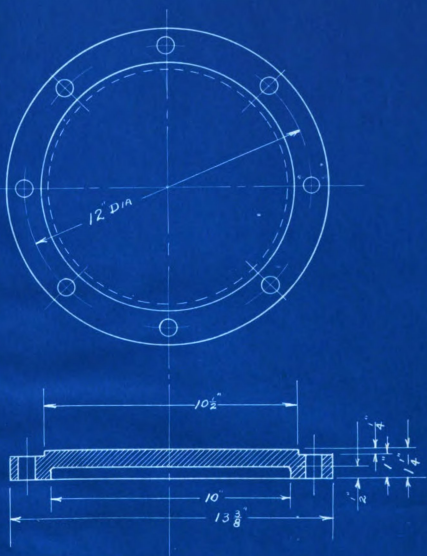
HANNA 30 T. RIVETER
VALVE
THESIS OF
H. MORTON NE. MARCH
M. A. C.
JUNE 1916



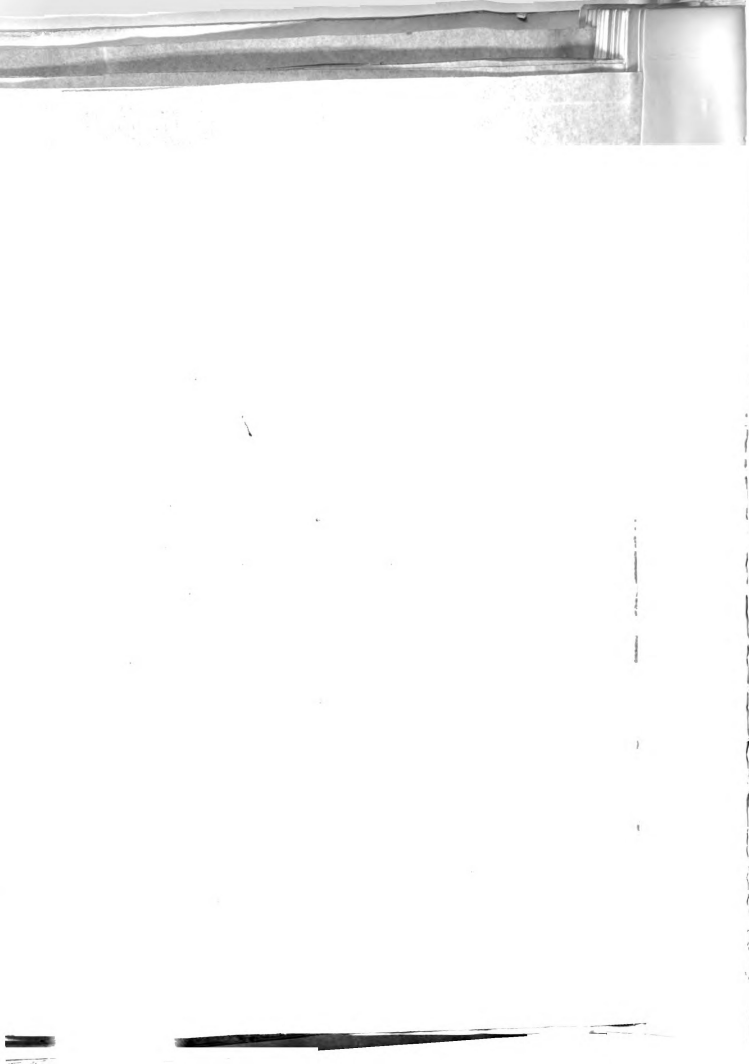


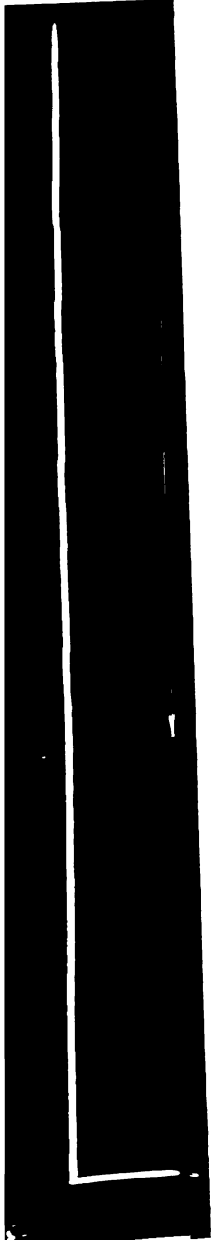


29



HANNA 30 T. RIVETER
CYLINDER HEAD
THESIS OF
H.E. MORTON N.E. MARCH
M.A.C.
JUNE 1916



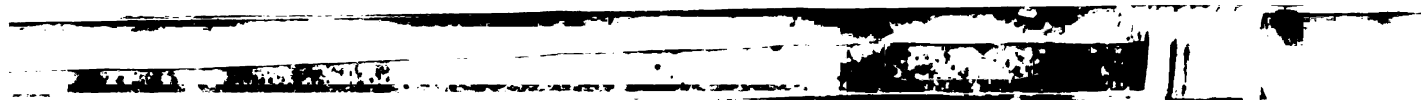












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